Fermilab

TM-1047
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A Summary of Satellite Refrigerator Dry Engine Efficiencies Calculated from Data Taken from January 1980 to April 1981

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INTRODUCTION

Figure 1 is a graph of data for Satellite Refrigerator dry engine performance which were taken at main ring locations A1, A2, A4, B1, and at the Switchyard Service Building. Dry engine inlet and exhaust temperatures and pressures were recorded daily by people in the Satellite Refrigerator Support Group as a routine check on engine performance. Also, as a part of data gathering efforts during tests and normal operation, data were recorded by people in the Tevatron Cryogenic Support Group and in the Switchyard Group who were operating these refrigerators.

Dry engine efficiencies were calculated from inlet and exhaust temperatures and pressures. The efficiency here is defined as the real change in enthalpy of the helium as estimated from inlet and exhaust conditions divided by the isentropic change in enthalpy. Typically the enthalpy in, enthalpy out, and isentropic enthalpy out would be found from the NBS helium tables or from a temperature-entropy diagram for helium. But using the fact that down to about 15°K and below 20 atm helium pretty well approximates an ideal gas, the approximation efficiency = $\eta = 1.52(0.98 - \frac{T_{out}}{T_{in}})$ was used. (See Figure 2 for the derivation of this.) This formula is based on an inlet pressure of 20 atm and exhaust pressure of 1.2 atm, which are the Doubler Design Report (1979) pressures. Typical inlet pressures were actually about 16 atm with inlet temperatures from 32 to 25°K, so efficiencies based on actual inlet pressures would be about 0.03 higher than the result given by the formula. Exhaust pressures ranged from the 1.2 atm Design Report figure to 1.5 atm, the latter of which results in the formula underestimating efficiency based on true exhaust pressure by about another 0.03.

All the points plotted in Figure 1 except those in a larger circle are calculated using $\eta = 1.52\ (0.98 - \frac{T_{out}}{T_{in}})$, i.e., efficiency based on 20 atm in and 1.2 atm out. The circled points are efficiencies calculated from the average of all inlet and exhaust temperatures and pressures for that engine and are calculated using the NBS helium tables. Hence, the circled point for each engine is essentially an average of all the other points plotted for that engine and is shifted up to account for deviations of actual inlet and exhaust line pressures from the Design Report numbers.

Because the operating temperatures of the dry engine are typically not in the vapor pressure thermometer range (see Figure 3), most of the data taken does not yield an efficiency. Thus, the number of efficiency points on the graph does not reflect the amount of data taken.

The scatter in the data for each engine may partly be due to non-steady-state readings. Exhaust temperatures lag changing inlet temperatures resulting in apparently higher or lower ΔT's through the engine. The efficiencies are calculated from these temperatures and hence may be over- or underestimated. Two possible solutions to this problem are to take enough readings to obtain a good average or, secondly, to take frequent readings (such as one each half hour) and use only data which are constant for three or more consecutive readings. In cases where we have enough data, the above two methods have yielded comparable results.

The following is a brief explanation of each run, starting from the bottom of the efficiency scale and working up.

GFD2 at B1

Although the first cold insert* at B1 in GFD2 (Gardner-Fermi dry #2) ran for 2,000 hours, it was frosted around the piston shaft seal for much of the run and had an average efficiency of 0.46 for the data plotted, 0.48 (the circled point) for the data using the actual average inlet pressure (16 atm) and the NBS tables. Note that this engine appears to have started in July at this level, a trend with time is not apparent.

The replacement of this cold insert with one equipped for measuring blowby past the piston rings helped a little, raising the average efficiency to 0.53 for the data plotted and 0.56 (the circled point) using the actual average inlet pressure (again 16 atm) and the NBS tables. Interestingly, the effect of the measured piston ring blowby on efficiency should have been only 0.02 on this engine. Apparently, ring blowby was not a major source of inefficiency here.

GFD2 at A1

TM-948 by T. Peterson and P. Brindza contains a detailed plot of efficiencies calculated from data taken both by Peterson and Brindza for this run. Data plotted here is from the engine logbook, recorded by various Satellite Refrigerator Support Group personnel and agrees well with that in TM-948. Efficiency averaged 0.55 for the first part of the run. It then dropped off to about 0.30 for a few days apparently

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*A cold insert is a replaceable subassembly consisting of the cold working parts: piston, shaft, shaft seal, cylinder, rings, cylinder head, cold rocker arms, valves, pushrods, and supporting spool piece.

due to contamination on the valve seats or piston rings. Warmup and a different intake cam (85^o instead of 100^o) then gave 0.65 efficiency. As at B1, there is no apparent trend in efficiency with time.

Perhaps the most encouraging result from the year of running GFD2 is that mechanically the warm end (all top end, room temperature parts) survived 4,000 hours of operation, approximately 96×10^6 cycles.

GFD3 at A1

Of about 25 readings at A1 for GFD3 only three are in the VPT range. The first two readings are on the same day and agree at 0.60 and 0.61. A mechanical problem in the cold rocker arms then forced a switch to a new cold end which yielded one data point on October 1 at 0.70.

GFD1 at Switchyard Service Building

A few readings from 1980 are plotted, ranging from 0.54 to 0.60. However, I have concentrated on the cold end which was installed December 20, 1980, since it has about 2,200 hours of operating time (at about 250-300 RPM, compared to the more typical 350-500 RPM for a satellite). All the points plotted are circled since each is an average of 5 to 10 points over a period of a day or a few days, and typical inlet pressures were 16 to 19 atm; exhaust pressures were about 1.2 atm.

The temperature and pressure data were taken by Switchyard people in the form of hard copies of the CRT readout. The large variations within these sets of data, as well as between them, is due to large variations of the indicated inlet temperature. The exhaust temperature and both pressures were much more consistent.

CTI-D-3 at B1

Daily temperature and pressure readings over the first 750 hours operation have yielded the 15 points plotted. However, the exhaust pressure averaged 7.5 psig and the inlet pressure averaged 203 psig, so the efficiency based on actual inlet and exhaust conditions is underestimated by the formula. The efficiency based on the actual average of the 15 sets of temperatures and pressures is 0.62, as indicated by the circled point on the graph. The average of the points plotted is 0.55.

GFD6 at A4

Claus Rode took readings every half-hour for a full shift while testing the A4 refrigerator. The resulting 15 efficiency points are plotted, ranging from 0.64 to 0.58 with a 0.62 average. However, as for CTI-D-3, pressures were not design pressures and result in the formula underestimating efficiency based on actual inlet and exhaust pressures. The average inlet pressure was 234 psig, and the average exhaust pressure was 7 psig. The efficiency based on these pressures and the recorded temperatures averaged 0.67 for the 15 readings, as indicated by the circled point on the graph.

CTI-D-1 at A2

The dry engine was colder than the VPT range for most of its 500-hour life, but 5 efficiency points were obtained ranging from 0.74 to 0.63 with an average of 0.67. Since exhaust pressures were a few pounds and inlet pressures averaged 230 psig, the average efficiency based on actual inlet and exhaust conditions is 0.69, indicated by the circled point on the graph.

GFD4 at A2

This engine ran for about 200 hours during September and was "frozen out", colder than VPT temperatures for most of the run. But on September 23, with about 150 hours of running time on the engine, in the course of doing transfer line studies Jay Theilacker took readings every half-hour for a full shift which yielded 7 efficiency points ranging from 0.60 to 0.68 and averaging 0.66. Exhaust pressures were a few pounds and inlet pressures averaged 195 psig, resulting in an average efficiency based on actual inlet and exhaust pressures of 0.70, indicated on the graph by the circled point.

Conclusions

1. The Gardner-Fermi engines are mechanically good for at least 4,000 hours of operation at 450 RPM, and the CTI dry expander, CTI-D-3 is at 1,000 hours at the time of this writing and still running.

2. Some improvements in Gardner-Fermi piston-shaft seal design, a new valve stem seal design, and larger "breather" holes to minimize pressure oscillations in the volume above the piston and below the shaft seal in GFD4, GFD6, and the new cold end in GFD1 at switchyard seem to have yielded efficiencies between 0.60 and 0.70 and have extended the piston shaft seal lifetime. In the case of GFD1 the engine has maintained these efficiencies for over 2,000 hours at around 300 RPM.

3. No trends of efficiency with time have yet appeared, although more data may show a downward trend at Switchyard. GFD2 never showed a clear trend at A1 or at B1. We have no other long-term data.

4. CTI dry engines also appear to have efficiencies between 0.60 and 0.70.

5. Data from B1 for both the Gardner-Fermi dry engine and the CTI dry engine had the most scatter and the lowest averages. This suggests a correlation between efficiency and the refrigerator in which the engine operates which might be the result of several factors. First, engine efficiencies depend on helium purity. Low efficiencies followed by spontaneous recovery may indicate contamination on valve seats causing temporary leaks. Secondly, engine efficiencies are better with operation at steady speeds and temperatures. Operation at one speed or temperature may wear seals and rings to dimensions not well suited for operation under other conditions due to frictional heating, thermal expansion, etc.

FIGURE 1

Tom Peterson
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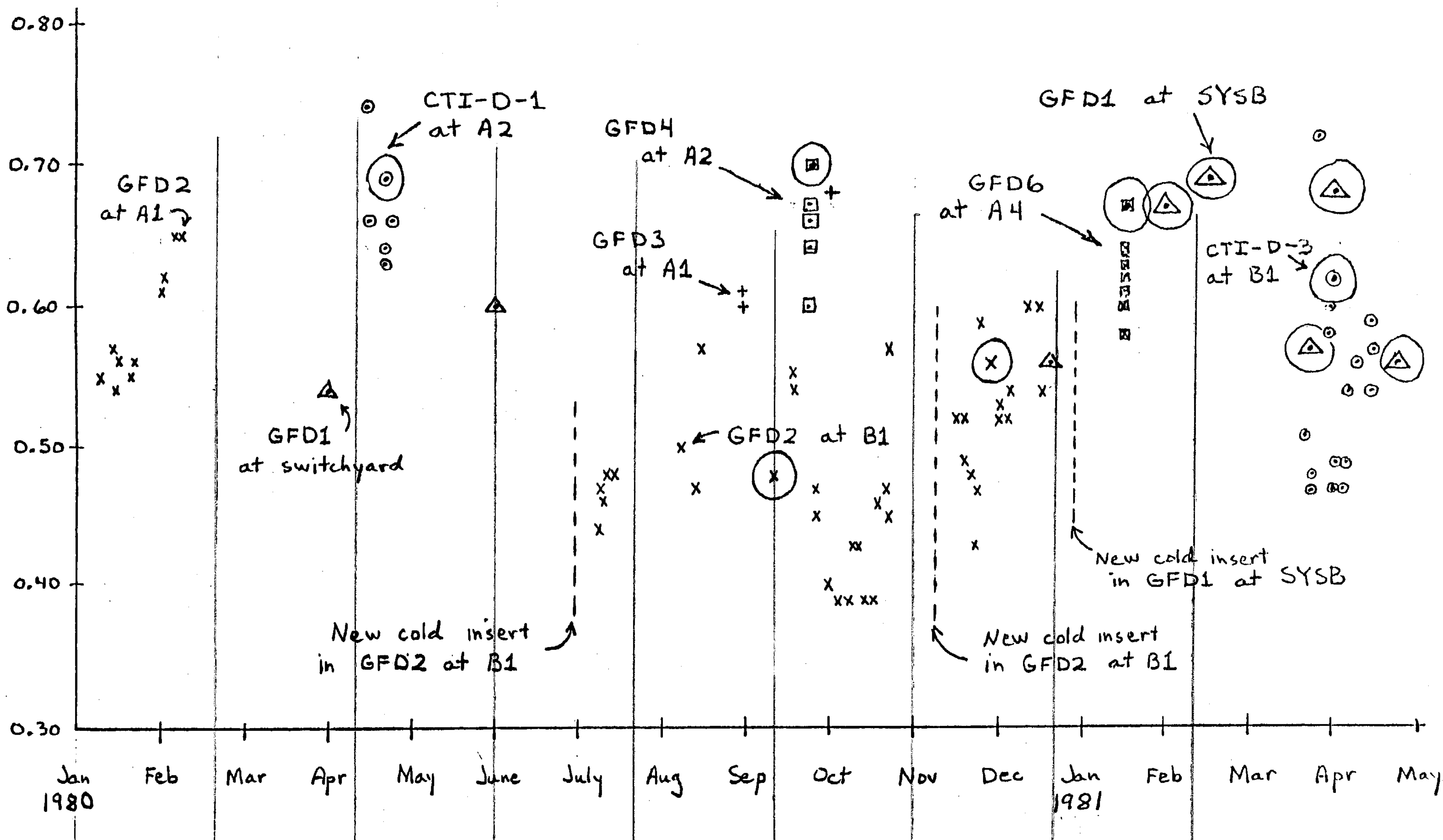
DRY ENGINE EFFICIENCIES



$$\eta \equiv 1.52\left(0.98 - \frac{T_{out}}{T_{in}}\right)$$

FIGURE 2

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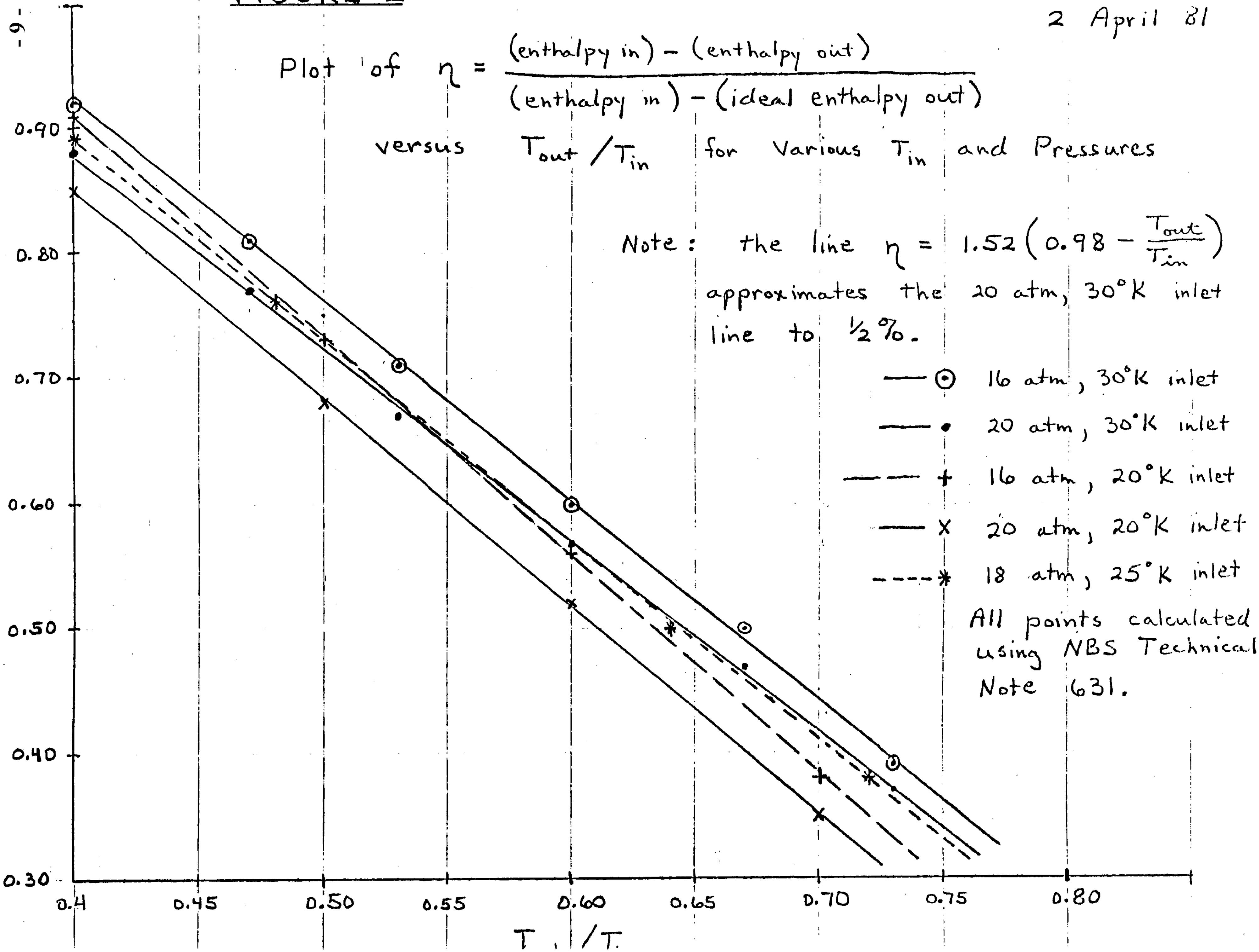

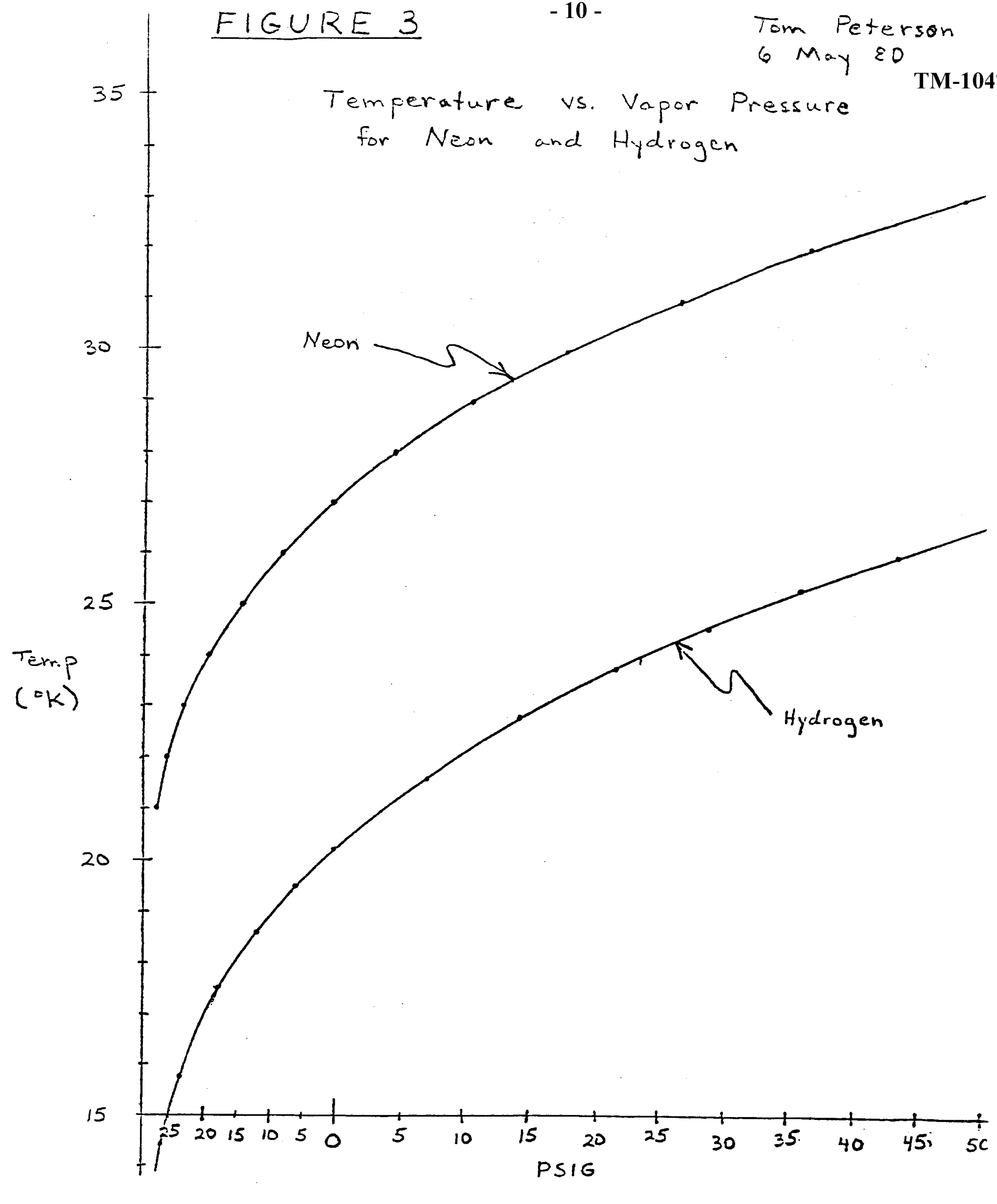
FIGURE 3
Tom Peterson
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Temperature vs. Vapor Pressure for Neon and Hydrogen
35
30
25
20
15
Temp (°K)
Neon
Hydrogen
25 20 15 10 5 0 5 10 15 20 25 30 35 40 45 50
PSIG